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Unclassified

INTERIM DEVELOPMENT REPORT

For

THE DEVELOPMENT OF GAS LUBRICATED BEARINGS FOR USE IN BLOWER MOTORS

This report covers the period 7 December 1962 to 7 March 1963

ROTRON

ROTRON MANUFACTURING CO.
Woodstock, N. Y.

NAVY DEPARTMENT BUREAU OF SHIPS ELECTRONICS DIVISION
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ABSTRACT

Start-stop testing on a prototype motor-blower unit demonstrated the ability of the gas bearing design to withstand 66,600 start-stop cycles without harm to the performance of the bearing. Continuous operation of the prototype unit for 1192 hours produced very little wear on the thrust bearing surfaces, indicating their ability to meet the 20,000 hour design life requirement of this development program.

Three units made to final design configuration successfully completed vibration and salt spray testing per MIL-STD-202B, Method 201A, with no detrimental effects on bearing performance. All testing to date indicates that the final design units will meet all performance and qualification test requirements of contract NObsr 87522.

Work has been completed on the section of a gas bearing design manual dealing with the prediction of half-frequency whirl, and work continues on the final section of the manual, which concerns the design of gas thrust bearings.

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I. PURPOSE

The purpose of the work described in this report is to develop gas lubricated journal and thrust bearings for use in blower motors. The specific goals to be accomplished are:

1. Establishment of theoretical design curves that can be used for the design of gas lubricated bearings suitable for use in the motors of cooling devices delivering air between 10 and 500 cfm.

2. Design and fabrication of a gas lubricated blower motor meeting the requirements of Military Specification MIL-B-21399 and Bureau of Ships Drawing RE-46-C-2105.

3. Demonstration of design capability through performance and qualification tests on three (3) blower units.

4. Delivery to the Bureau of Ships of six gas-bearing blower motors and one set of detailed manufacturing drawings.

To accomplish these goals the program has been broken down into four phases as depicted on the chart on Page 22.

In the first phase bearing design curves will be established using available theoretical data. Wherever possible the design curves will be checked by experimental data.

During the second phase an actual gas-bearing blower motor in the 20 cfm range will be designed and fabricated. Aerodynamic and electrical performance characteristics will be established prior to beginning phase three qualification testing. In the final phase of the program six gas bearing blower units and the associated drawings will be shipped to the Bureau of Ships for further evaluation.

II. GENERAL FACTUAL DATA

A. Personnel

The work presented in this report is being conducted at the Rotron Manufacturing Company with Mr. D. S. Wilson acting as project engineer. Mr. D. D. Fuller (Professor, Columbia University) is serving in the capacity of gas bearing consultant and is responsible for establishment of the theoretical design curves. Electrical design is under the cognizance of the Chief Electrical Engineer, Mr. J. Ebbs, with Mr. B. Larys of the Rotron Research Corporation acting as electrical consultant. The mechanical design is under the cognizance of Mr. D. Harris, the Chief Designer. Mr. R. East is acting as project technician for this program. The accumulated hours spent during this report period by the above personnel are summarized as follows:

D. S. Wilson	- 597
D. D. Fuller	- 189-5/12
J. Ebbs	- 42
B. Larys	- 40
D. Harris	- 27
R. East	- 360

B. References

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III. DETAILED FACTUAL DATA

A. Theoretical Design Curves

Work is continuing on a Design Manual which presents theoretical design curves for use in designing gas bearings for blower motors. Work has been completed covering load capacity, friction, synchronous whirl and half-frequency whirl for plane cylindrical bearings. The last section is presently being completed covering the design of gas thrust bearings.

Considerable effort has been expended in preparing the section of the manual concerned with half-frequency whirl. Three approaches of this type of instability are presented in the manual. The first approach is presented to approximate the threshold of translatory half-frequency whirl for an oil lubricated bearing utilizing the following derived [25, 26] expression:

$$\frac{f_{TS}}{2} = \frac{1}{\pi}\sqrt{\frac{1}{\frac{ms}{2}\left[\frac{1}{k_1}+\frac{1}{k_2}\right]}} \qquad (1)$$

where: $\frac{f_{TS}}{2}$ = shaft speed at which half-frequency translatory shaft whirl begins (cycles per sec.).

ms = mass of shaft-rotor system (lb-sec^2/in).

k_1 = shaft spring stiffness (lb/in).

k_2 = radial spring stiffness of the bearing film for one bearing (lb/in).

In those applications where the shaft system operation is well below the first critical speed, the shaft system stiffness is much greater than the radial stiffness of the gas film and can be neglected without introducing a significant error and equation (1) becomes:

$$f_{TS} = \frac{1}{\pi}\sqrt{\frac{2k_2}{ms}}$$

This expression has been used for compressible bearing fluids. However, with increasing bearing compressibility numbers (Λ) the error increases significantly. Figure 1 plotted from the data in Table 1 presents the ratio "C" of predicted threshold value to measured threshold value for two different length to diameter ratios.

For bearings with short length to diameter ratios, it is possible to predict the ratio "C" through the use of the stability parameter [33, 34] ω_1 which is defined as:

$$\omega_{\mathrm{I}}^{*} = \omega\sqrt{\frac{cM}{w}}$$

Where: ω = shaft speed (radians/sec)
c = bearing radial clearance (inch)
M = rotating mass per unit length of bearing ($lb\text{-}sec^2/in^2$)
w = bearing load per unit length of bearing (lb/in)

This parameter has been included in Table 1 and can be cross-plotted with Λ and "C" to produce the stability data curves for a length to diameter ratio of 1/2 as illustrated in Figure 2. This approach, however, is limited to length to diameter ratios to approximately 2.5. Above this 1/d ratio the approach presented by Professor V. Castelli at Columbia University (presently being prepared for a Doctoral Thesis) was followed. In this case, the compressibility number Λ is plotted against the stability parameter ω_{I}^{*} for various eccentricity ratios (ε_0) for an infinitely long bearing. Figure 3 presents a plot of stability curves following this approach. To utilize this curve it is necessary to determine the steady state eccentricity ratio of the bearing using the infinitely long bearing load analysis and plot this point versus the stability number and compressibility number. If the point falls to the left of the stability line for that eccentricity ratio, bearing operation will be stable.

Table I

Calculated & Observed Threshold Speeds

No.	Source Ref.	l in.	d in.	c in.	Observed Threshold Rad/Sec	Λ	Predicted Threshold Rad/Sec	Ratio "C"	ω_{I}^{*}
1	(33)	0.5	1.0	0.000533	584	0.549	1765	3.02	0.745
2	(33)	0.5	1.0	0.000284	572	1.89	2830	4.94	0.498
3	(33)	0.5	1.0	0.001175	584	0.113	844	1.44	1.435
4	(33)	0.5	1.0	0.000284	402	1.33	2040	5.07	0.416
5	(33)	0.5	1.0	0.000533	383	0.360	900	2.35	0.646
6	(33)	0.5	1.0	0.000533	218.9	0.206	427	1.94	0.513
7	(34)	2.5	2.5	0.001250	680	0.736	1338	1.965	1.312
8	(34)	2.5	2.5	0.001250	560	0.604	1100	1.97	1.195
9	(34)	2.5	2.5	0.001250	440	0.475	684	1.55	1.115
10	(34)	2.5	2.5	0.001250	785	0.847	1557	1.985	1.410
11	(32)	2.0	2.0	0.001643	1235	0.504	1840	1.49	2.55
12	(32)	2.0	2.0	0.001643	1095	0.455	1595	1.39	2.42

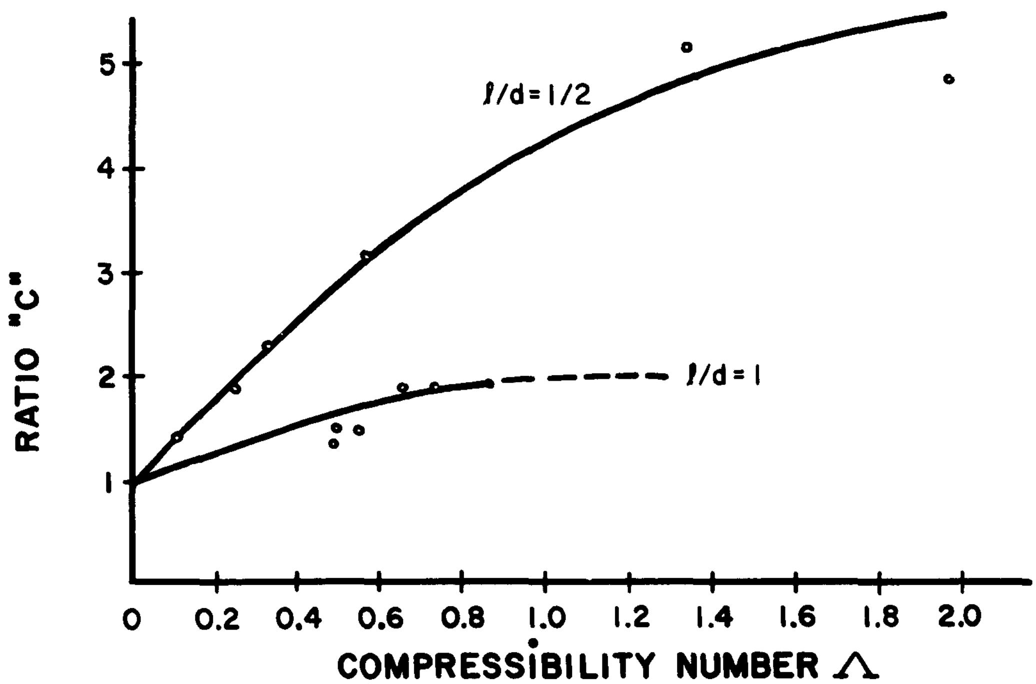

Figure 1
Ratio "C" vs Compressibility Number for l /d Ratios 1 and ½

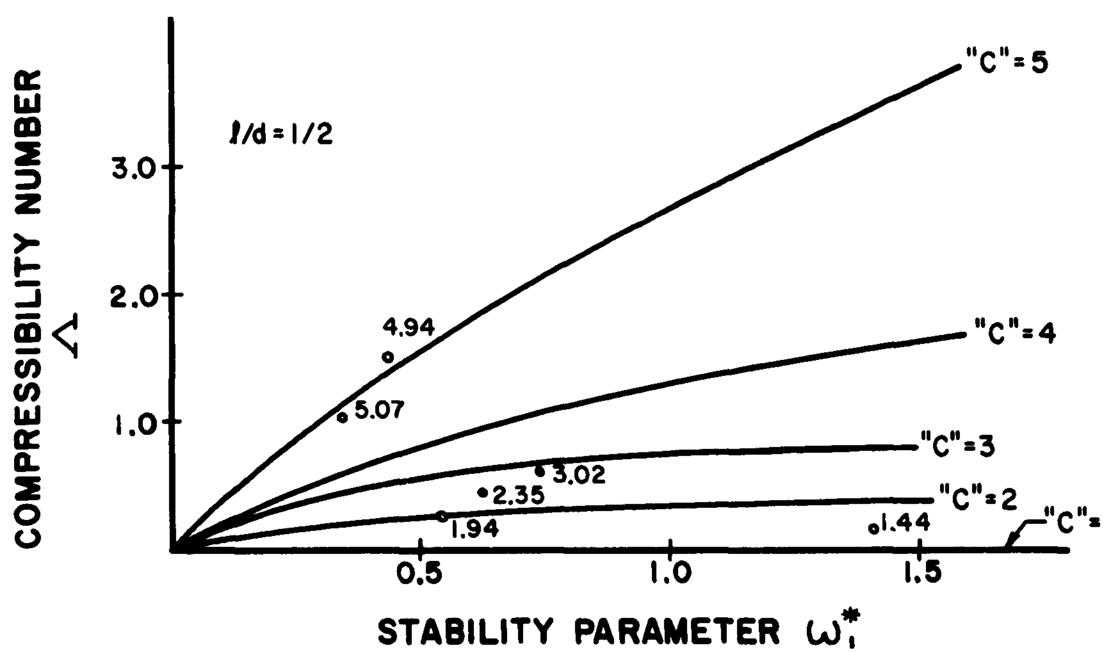

Figure 2
"C" vs Λ and ω_1^* for l /d ½

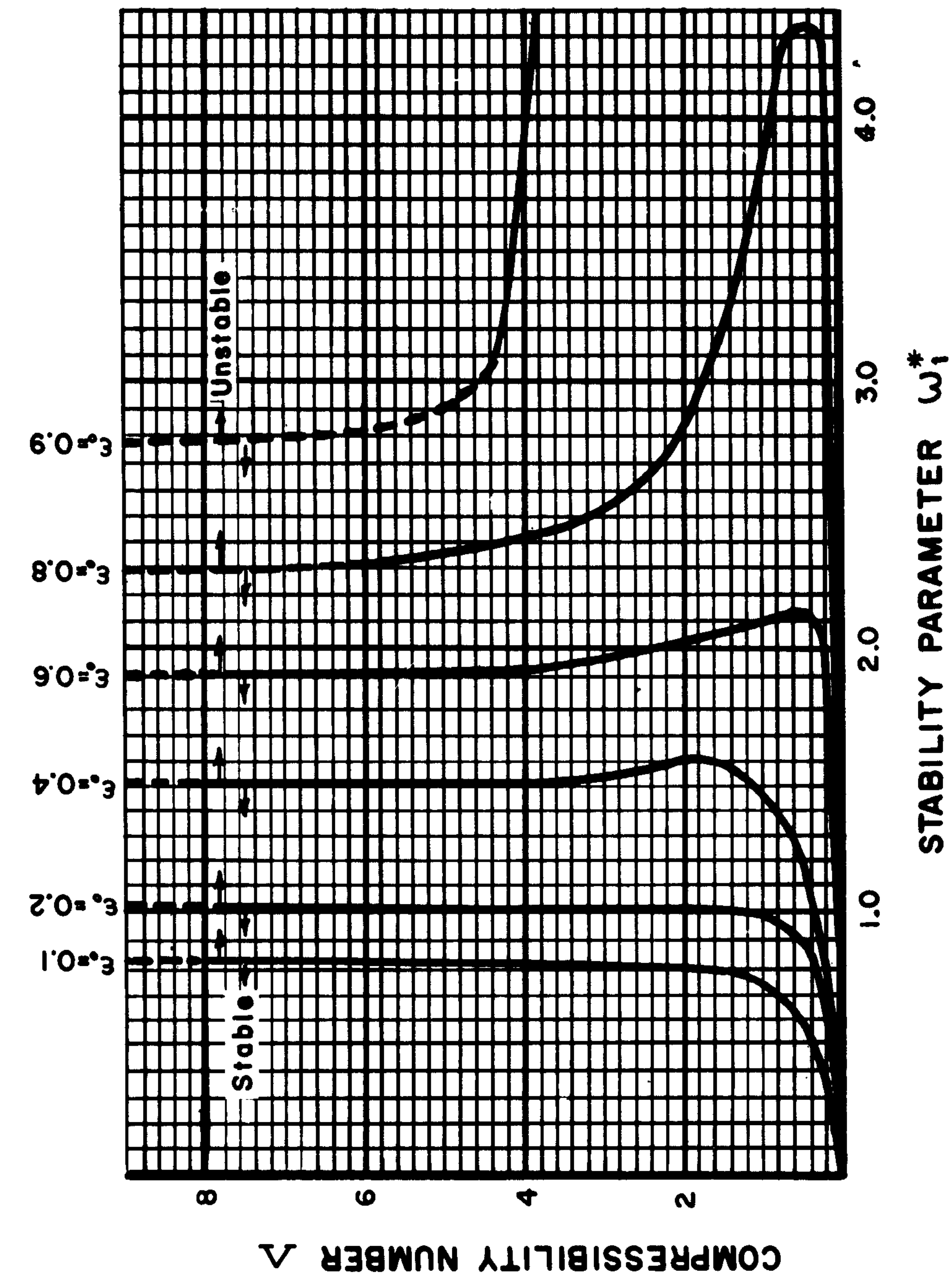

Figure 3
Stability Map for Infinitely Long Bearings

B. Test Rig Evaluations

Several stability tests were conducted in the test rig to check the analytical approach for determination of the threshold of half-frequency whirl. The procedure utilized to determine experimentally the threshold speed is outlined as follows:

The test rotor was installed in the rig instrumented with four capacitive probes as described in references 29 and 30. The x-y probes (located 90° apart at one end of the rotor) were fed through a Wayne-Kerr capacity meter to the horizontal and vertical channels of a Tektronix type 502 Oscilloscope. As the rotor was rotated a circular orbit was described on the oscilloscope screen which depicted the run-out or eccentricity of the rotor circumference passing under the probe to the center of rotation of the rotor. This eccentricity has been maintained below 100 microinches for these tests. As the rotor became airborne, any unbalance in the rotor caused an additional circular input at synchronous speed to the oscilloscope trace. The resultant trace was a vector sum of both unbalance and run-out and depended upon their phase relation to determine the actual distortion that would be evidenced on the trace. During the low speed evaluations (below 4,000 rpm) the unbalance component was too small to be observed. An index mark was provided on the rotor as a reference to distinguish each rotation of the rotor.

As rotor speed was increased, the first tendency for half-frequency whirl could be noted by the change of the circular orbit on the scope into two concentric circles. The speed at which this occurred was recorded as the instability threshold speed. Further increase in speed caused the inner circle to decrease in size and the outer circle to grow as depicted on Figure 4.

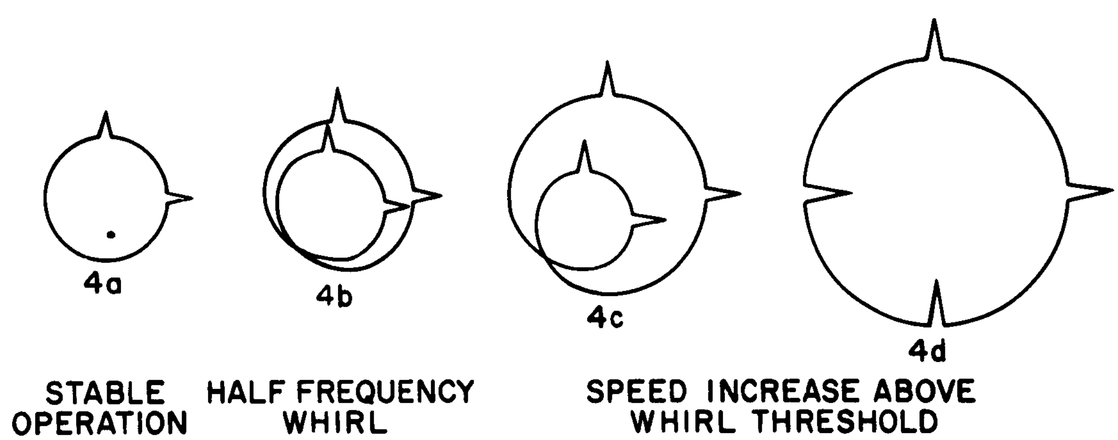

Figure 4
Rotor Center Whirl Paths Through Half-Frequency

With decreasing speed, stability was regained at a lower frequency than during increasing speed, which can be seen in Table 1 as the first of the two numbers under threshold speed. When instability was encountered, the mode of whirling was determined by observing the sine wave traces of the vertical capacitive pick-ups at either end of the rotor simultaneously through the vertical input channels of the dual beam scope. Figure 5 illustrates the change in wave form and phase relation observed on the scope when whirl is encountered.

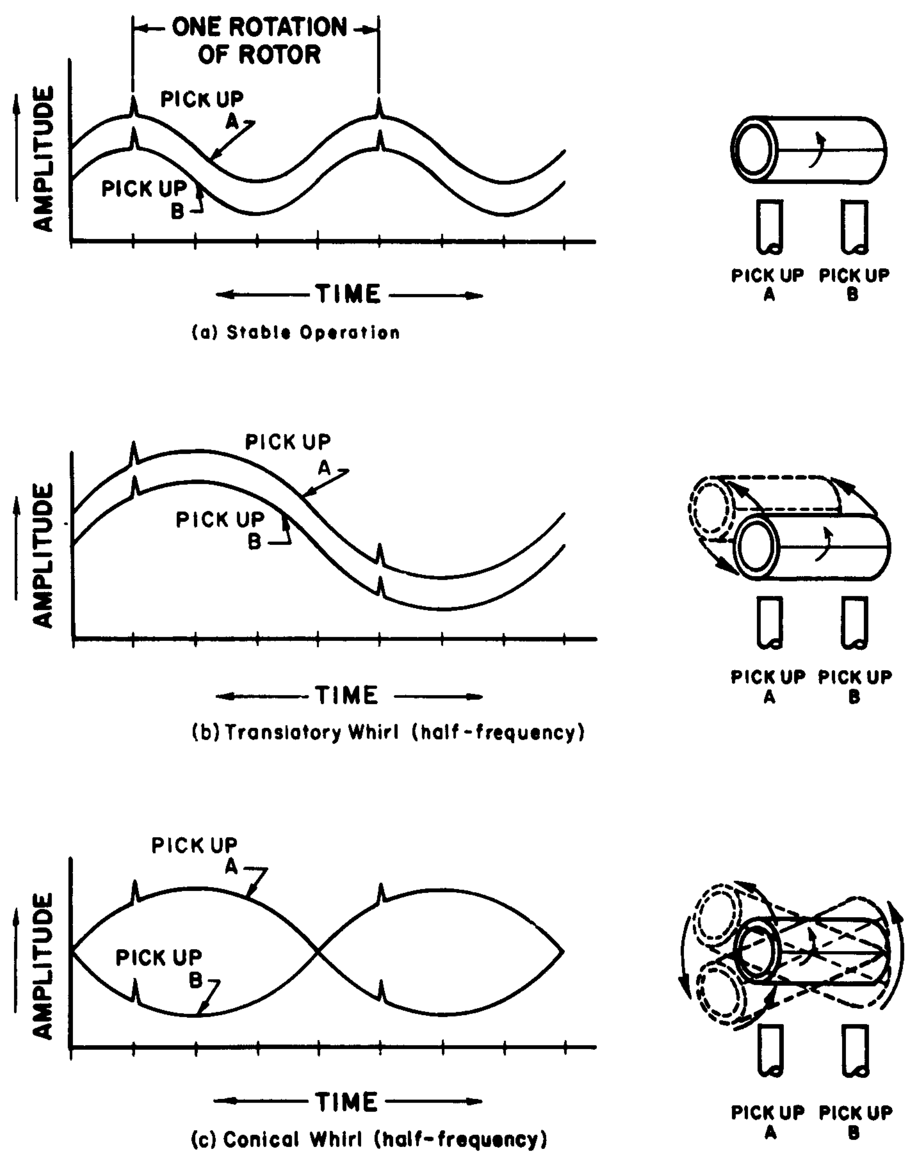

Figure 5
Whirl Modes as Observed on Oscilloscope

The results of the tests conducted are presented in Table 2 and plotted on Figure 3. To determine whether the air drive jets might influence the observed point of half-frequency whirl, the point at which stability was regained was determined by decreasing speed by shutting off the air supply completely, and by reducing the air supply slowly. These tests showed no difference in the results, and it was therefore concluded that the air jets were not affecting the observations. As the threshold was approached, however, any slight shock or disturbance would induce whirl immediately.

As noted on Figure 6, the whirl threshold speed increased as the diametral clearance was increased above the observed critical minimum.

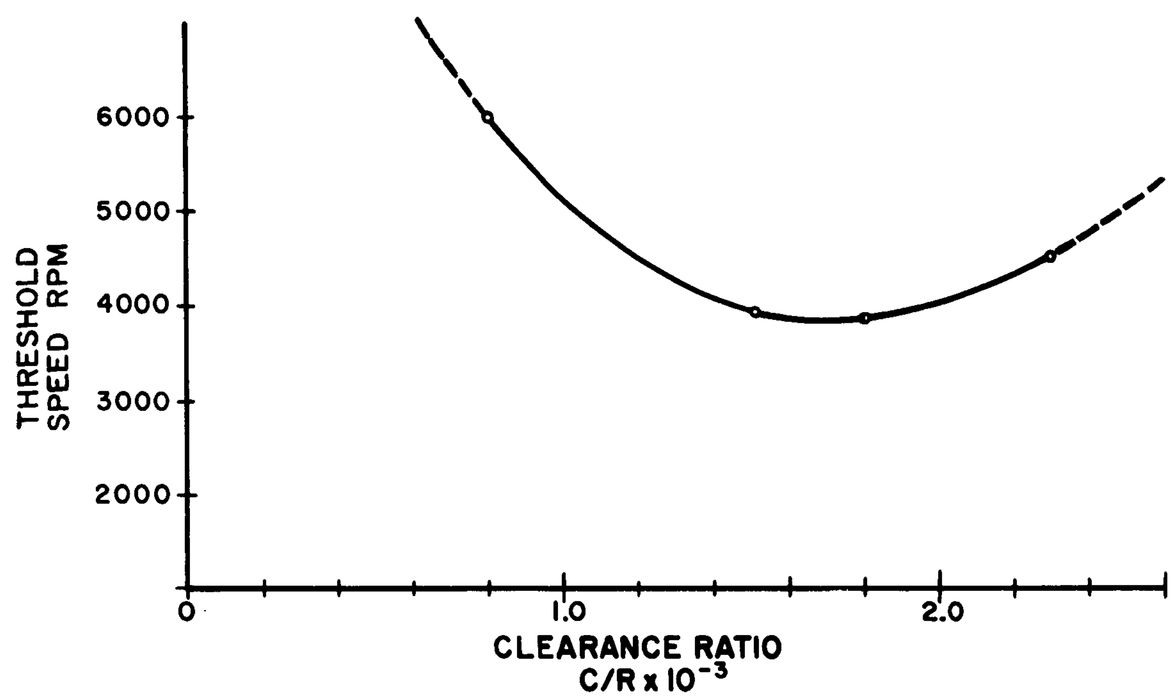

Figure 6
Plot of Half Frequency Whirl Threshold
For 3/8 inch Bearing (ℓ /d=3.33)

Table 2

Half-Frequency Whirl Threshold

Rotor Wgt.	c	L/D	Threshold Speed-RPM	Λ	ε Calc.	Type of Whirl	$\frac{c}{R} \times 10^{-3}$	Rotor
143 gr.	.000153	3.33	5750/6000	1.033	0.04	Conical	0.815	A
155	.000293	3.33	3820/3960	0.183	0.23	Conical	1.562	B
155	.00034	3.33	3810/3900	0.1325	0.31	Conical	1.814	B
155	.00043	3.33	4400/4500	0.0970	0.41	Conical	2.29	B

C. Gas Bearing Blower Motor

The initial development blower unit was fabricated utilizing a 3/8 inch diameter ceramic bearing sleeve 1-1/4 inches in length. The blower wheel on this unit was removable, which permitted versatility during the development phase of the program. In contrast, the final qualification units contained 0.420 diameter ceramic bearings 1-1/4" in length and blower wheels permanently attached to the rotor assembly to facilitate balancing.

Two vibration tests and a start-stop test were conducted on the development unit as outlined in Table 3.

Table 3

Development Tests Conducted on Prototype Blower

Bearing Radial Clearance: 125 micro inches
1/d Ratio: 3.33
Bearing Diameter: 0.375 inch
Rotor Weight: 0.26 lbs.

Date	Test	Remarks
12-17-62	Horizontal Operation	120 hours @ 22°C ambient
1-3-63	Vibration	0 to 55 cps scan at 0.06 double amplitude in horizontal plane-satisfactory. 0 to 55 cps scan at 0.06 double amplitude in vertical plane-bearing hit at 27 to 30 cps.
1-4-63	Horizontal Operation	178 hours-stopped to install thrust bearing preload spring.
1-17-63	Horizontal Operation	122 hours @ 22°C ambient.
1-23-63	Vibration Test Repeated	Bearing touched at 25 to 30 cps when mounted in vertical plane.
1-23-63	Horizontal Operation	116 hours @ 22°C ambient.
1-28-63	Start-Stop Test	66,600 start-stops in horizontal plane. Coast down time remained constant at 9 seconds throughout test.
2-11-63	Horizontal Operation	652 hours - stopped to inspect unit.

Total running time - 1192 hours.

During vibration in the vertical plane bearing contact could be audibly distinguished at 25 to 30 cps vibrational frequency. The applied frequency at the point of contact was approximately one half unit frequency. It has been shown in the past [31] that bearing load capacity is reduced considerably at a ratio of applied vibrational frequency to rotational frequency of 0.5. Because of the low load level at the applied frequency it was not felt that the reduced load capacity was the only cause for bearing contact. During vibration in the vertical direction contact occurs between the mechanical thrust washers, which can produce an additional side load component on the bearing if the washers are not square to the bearing centerline. An unsuccessful attempt was made to eliminate this condition by spring loading the thrust washer. The magnitude of the thrust force, however, was sufficient to collapse the light load spring and cause contact between the bearing journal surfaces at the 0.5 applied frequency to operating frequency ratio. In contrast, however, contact was not evidenced at this point with the 0.420 diameter bearing on three qualification units that were vibrated. Thrust washer alignment was considered superior in this configuration because the rotating washer was not rigidly secured to the rotor, creating a self aligning feature.

A start-stop test was conducted on the development test unit for a total of 66,600 start-stop cycles (see Table 3). This test consisted of energizing the blower motor for a period of two seconds every thirty seconds. The unit was not disassembled after this test but was continuously operated for 652 hours immediately following the start-stop evaluation. After a total running time of 1192 hours, the unit was disassembled for inspection. The condition of the thrust surfaces after this operation can be seen in Figure 7. There were no signs of excessive wear and it appears that the unit will be capable of 20,000 hours of operation on these surfaces without difficulty. The wear area on the duroid material washer that contacts the crowned end of the shaft was caused primarily during vibration test due to end bump. This condition was also noted on the qualification test units after vibration in the shaft vertical position. Average increase in endplay was 0.015 inches after this test. There were no detrimental bearing performance effects noted from this condition and it was therefore considered acceptable.

Three qualification units were assembled utilizing 0.420 inch diameter ceramic bearings 1-1/4" long for qualification testing. Table 4 summarizes the testing conducted to date on these units.

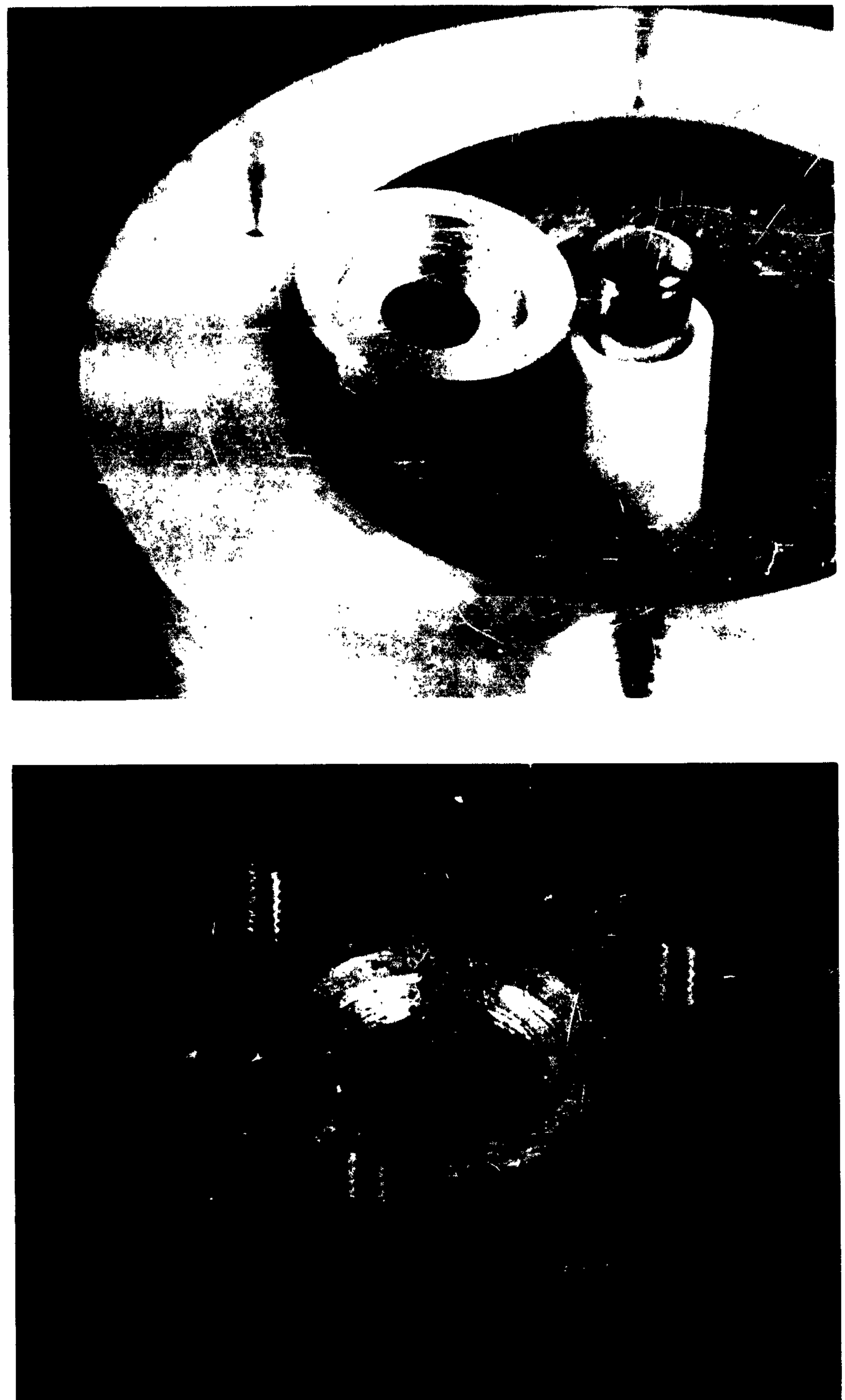

Figure 7 – Thrust Bearing Surfaces After 1192 Hours of Operation

TABLE 4

QUALIFICATION TEST RESULTS

Unit No.	Average Bearing Radial Clearance	Type Test	Results
1	0.00017	Vibration Test per MIL-STD-202B Method 201A (0.06" double amplitude: 10 to 55 cps to 10 cps in 1 minute in three mutually perpendicular planes)	No bearing contact during test. Unit starts and stops satisfactorily after test.
		Salt Spray Test at 33°C for 48 hours per MIL-STD-202B Method 201A	Unit started and stopped 3 times after test satisfactorily. Winding developed a leak to ground after test.
2	0.00024	Vibration Test as above	Bearing contact audible at 55 cps with vibration parallel to shaft axis. Unit started and stopped satisfactorily after test.
		Salt Spray Test as above	Unit started and stopped 3 times after test.
3	0.00026	Vibration Test as above	Bearing contact audible at 55 cps with vibration parallel to shaft axis. Unit started and stopped satisfactorily after test.
		Salt Spray Test as above	Unit stopped and started satisfactorily after test. Stator developed a leak to ground after test.

Figures 8, 9, and 10 depict the test setup and vibrational planes utilized for the vibration test. All three units operated satisfactorily throughout the test and started and stopped with no noted change in coast down time after vibration in each plane. A decrease in speed was noted on each unit at the peak load of 10 g's and 55 cps. The average speed drop was 200 rpm when vibrated in the plane parallel to the shaft axis and 100 rpm in the plane perpendicular to the bearing axis. Bearing contact was noted during vibration in plane B at the maximum load condition of 10 g's in two units containing average radial clearances above 0.0002 inches. A maximum radial clearance of 0.0002 inches has been specified for production units although it was necessary to utilize the greater clearance bearings for qualification due to procurement difficulties in obtaining the desired clearance. An average of two to three contacts were made each vibrational sweep cycle or approximately 240 to 360 contacts of the bearing in the B plane of vibration. Despite this, the unit performed satisfactorily after the test with no evidence of performance decay. Visual inspection of the bearing showed no signs of damage after test and therefore the test was considered a satisfactory demonstration of the ability of the bearing and blower design to withstand the vibrational requirements set forth in MIL-STD-202B Method 201A.

The units were not disassembled following the vibrational test but rather subjected to a 48 hour salt spray test per Method 201A of MIL-STD-202B. The units were not operated during this test and upon completion were subjected to three start-stop tests prior to cleaning or washing the salt crust. Figures 11 and 12 depict the units after salt spray operation.

All three units operated satisfactorily after the salt spray test. However, after 24 hours, two units developed grounds in the stator winding. A review of the impregnation procedure indicated that the stators were inadvertently impregnated to the Rotron commercial standard in lieu of the military impregnation procedure. The stators will be replaced and the test repeated with properly impregnated stators. However, most importantly, the test demonstrated that the bearing is correctly designed to withstand the effects of salt spray. The test is therefore considered successful for the purpose of this development program.

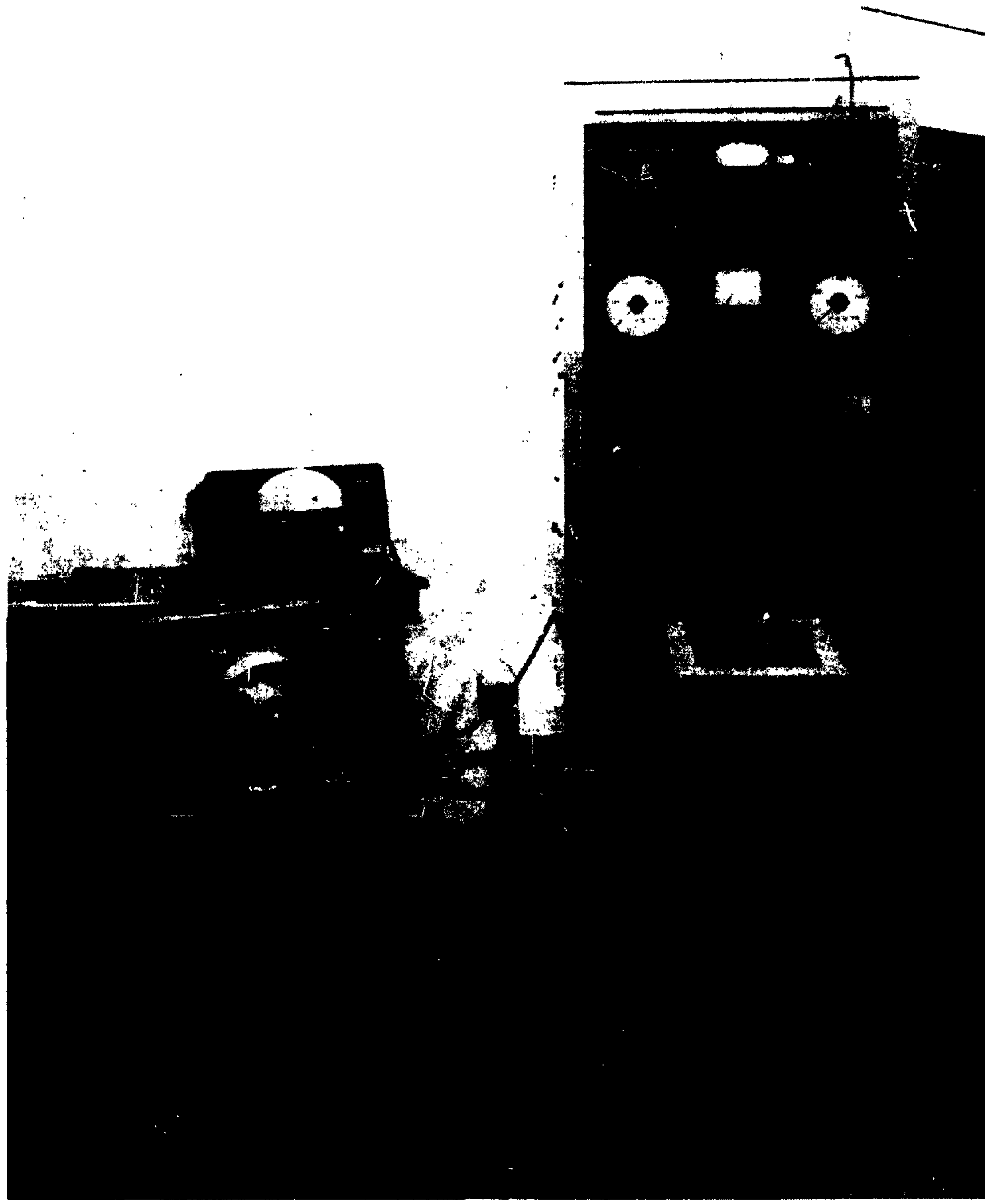

Figure 8 – Vibration Test Set Up Plane "A"

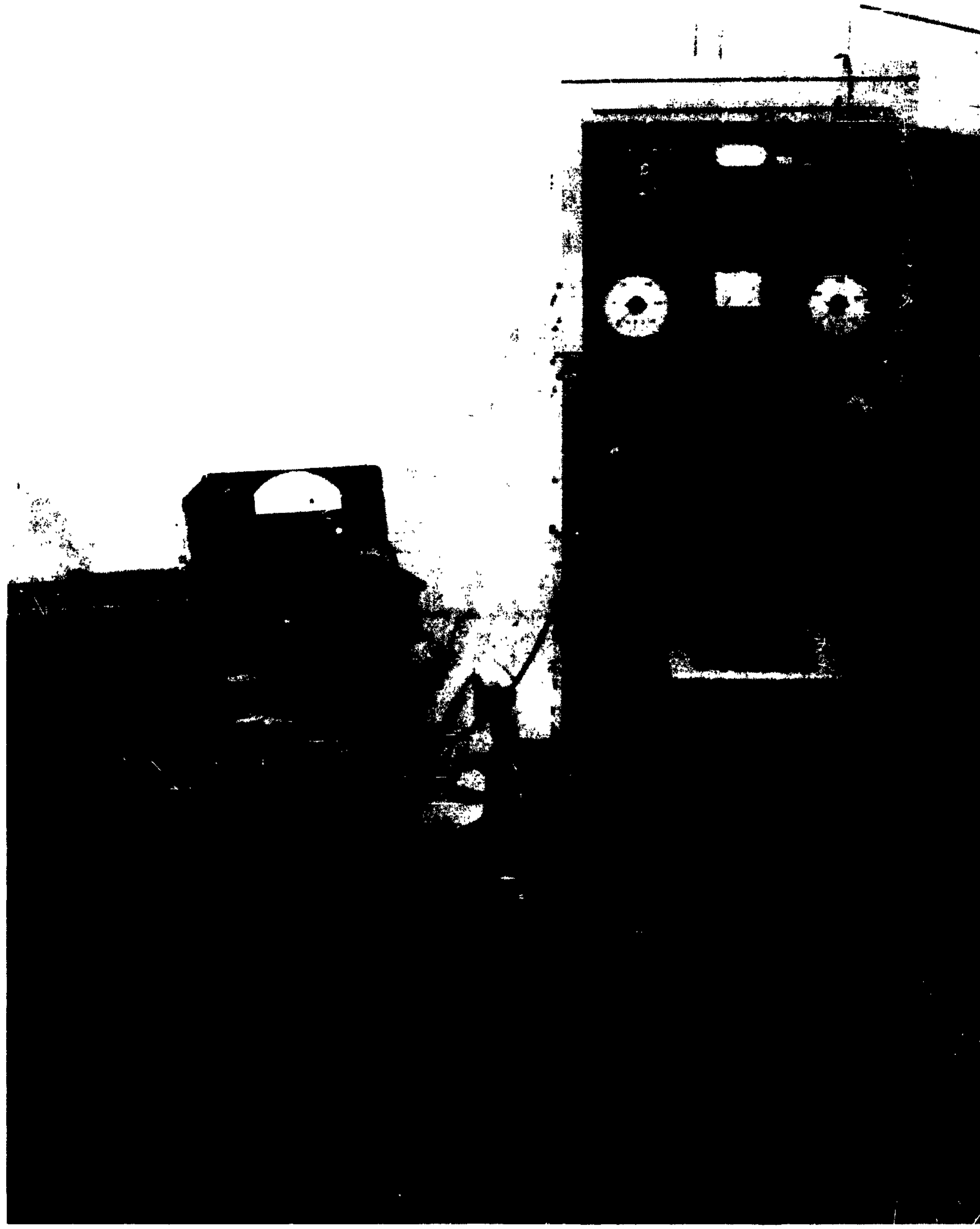

Figure 9 – Vibration Test Set Up Plane "B"

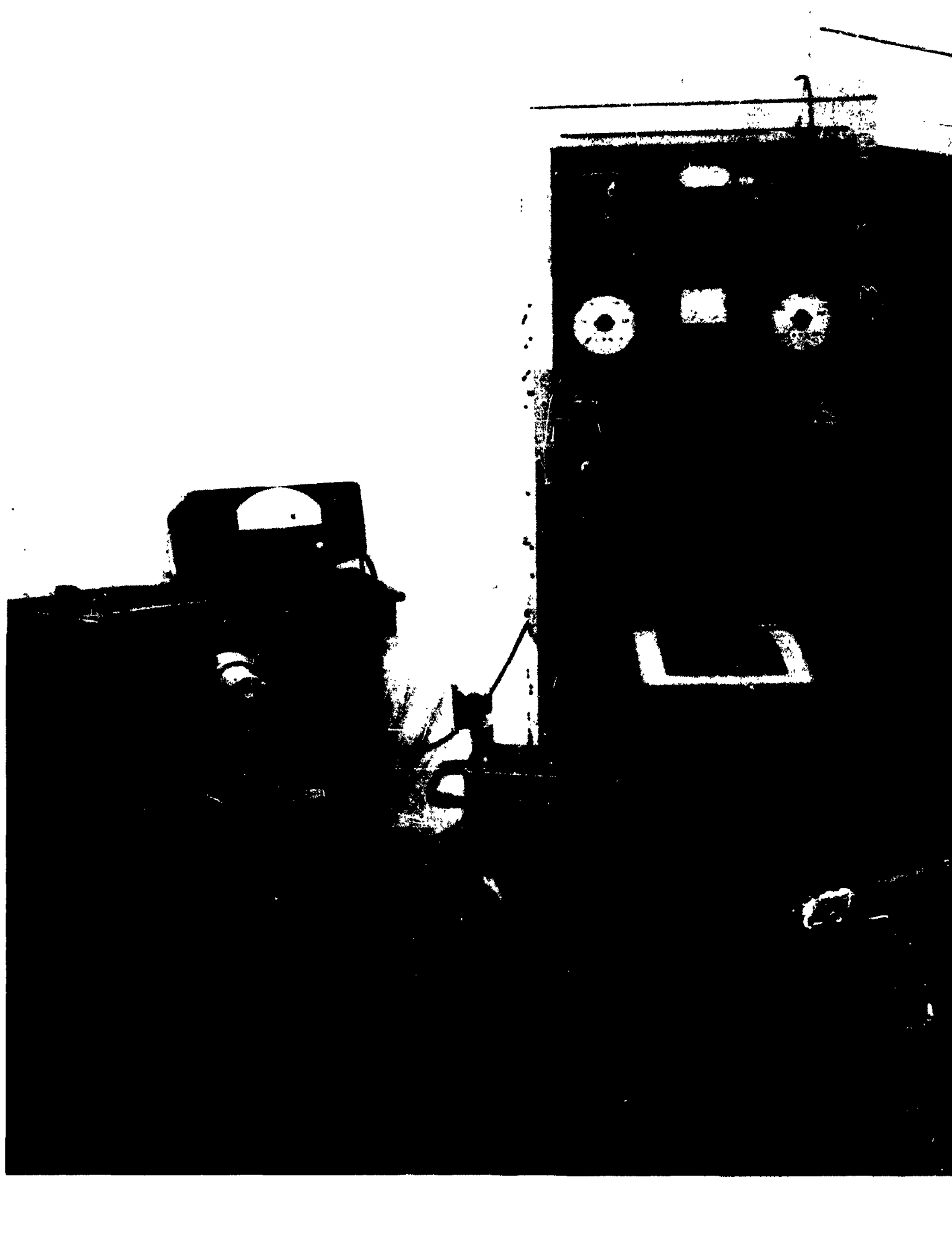

Figure 10 – Vibration Test Set Up Plane "C"

Figure 11
Qualification Units Installed in Salt Spray Chamber

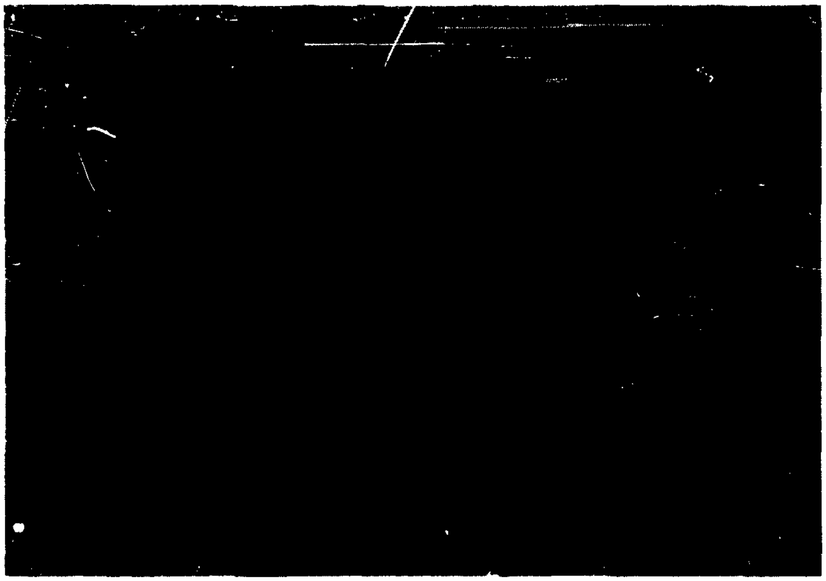

Figure 12
Qualification Unit After Salt Spray Test

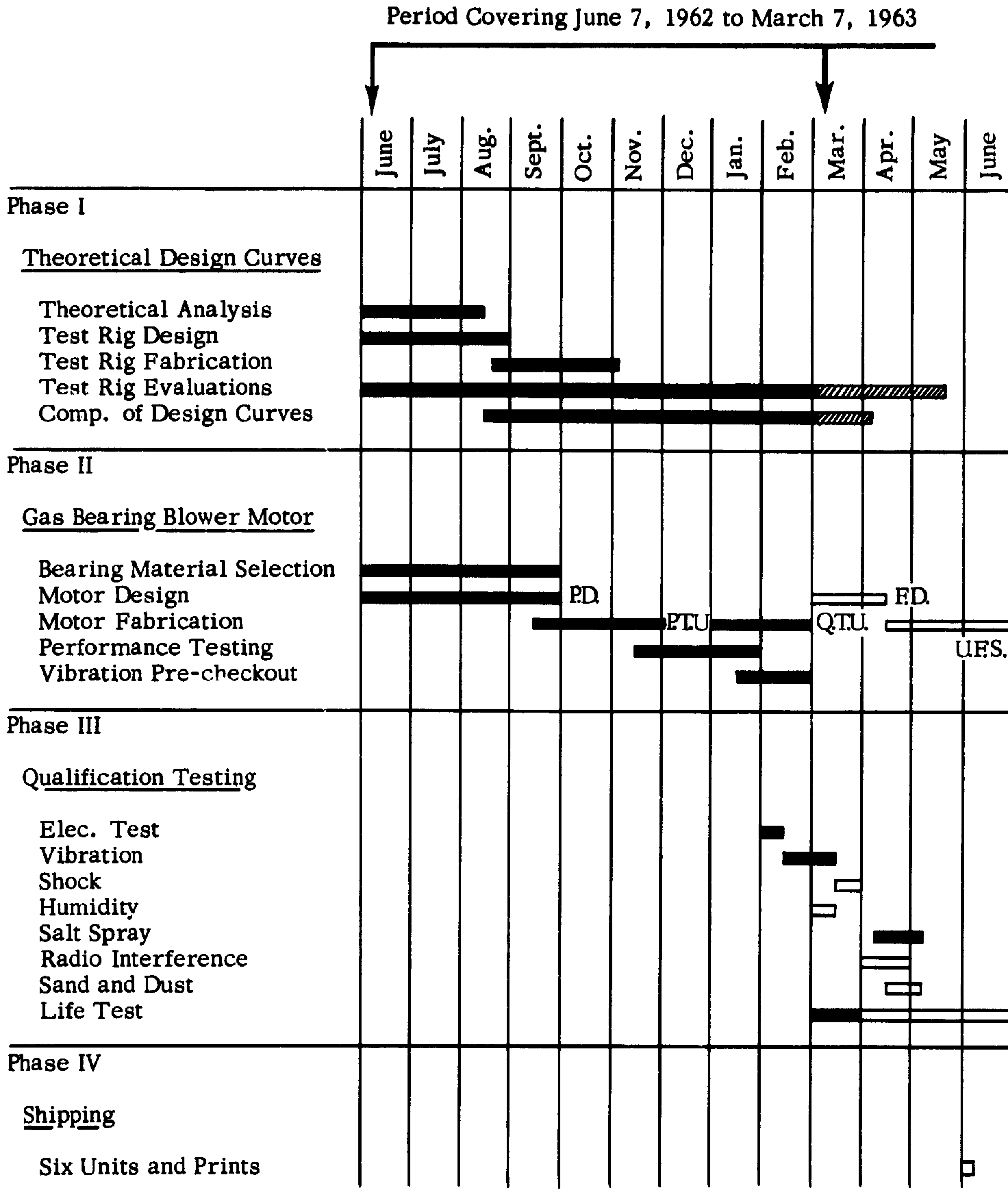

Figure 13 – Project Performance and Schedule
Rotron Manufacturing Co., Inc.
Contract No. NObsr - 87522

ROTRON MANUFACTURING COMPANY

PROJECT AND PERFORMANCE SCHEDULE

Legend	Description
	Work Performed
	Schedule of Projected Operation
	Revised Schedule of Projected Operation
P. D.	Preliminary Drawings
F. D.	Final Drawings
P. T. U.	Performance Test Units
Q. T. U.	Qualification Test Units
U. F. S.	Units for Shipment

Estimated completion in percent of Total Effort expected to be expended

1.	Initial Theoretical Analysis	100%
2.	Test Rig Design and Fabrication	100%
3.	Test Rig Evaluations	80%
4.	Gas Bearing Design Manual	90%
5.	Gas Bearing Blower Motor	85%
6.	Qualification Testing	10%

Notes

Test rig evaluations have been extended from 1/31/63 to 5/15/63. The completion of design curves has been extended from 1/31/63 to 4/7/63. All remaining phases of the program are on schedule.

CONCLUSION

Considerable effort has been expended on the section of the gas bearing design manual dealing with prediction of half-frequency whirl. Three approaches are being utilized for the theoretical determination of half-frequency whirl. The first approach can be utilized with oil lubricated bearings and very low compressibility number gas bearings. The second approach introduces a correction factor to correct the stability data for non-compressible flow for higher compressibility numbers and length to diameter ratios to approximately 2.5. The third approach is utilized for large length to diameter ratios.

Start-stop and endurance evaluations have been conducted on a development prototype blower-motor assembly. The unit has successfully operated for 66,600 start-stop cycles with no decay in bearing performance or coast down time. Continuous operation for 1192 hours including vibration testing has indicated very little wear of the mechanical thrust surfaces. It appears that the thrust surfaces will be quite capable of 20,000 hours of operating life.

The qualification phase of the program has commenced on three blower-motor assemblies representing the final configuration. All units successfully completed a six hour vibration test per MIL-STD-202B. Bearing contact could be distinguished on two units during vibration in the vertical plane. However, there appeared to be no adverse effects or bearing damage. The gas bearings also performed satisfactorily after a 48 hour salt spray test per MIL-STD-202B. There is no evidence to date to indicate that the units will not be capable of meeting all of the qualification test requirements set forth in NObsr-87522.